84-1568

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ALEXANDER L. STEVAS
CLERK

No.

IN THE

SUPREME COURT OF THE UNITED STATES

October Term, 1984

QUAKER CITY GEAR WORKS, INC., *et al.*,
Petitioners

v.

SKIL CORPORATION,
Respondent

APPENDIX B TO PETITION FOR A WRIT OF CERTIORARI TO THE UNITED STATES COURT OF APPEALS FOR THE FEDERAL CIRCUIT

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INVOLUTE GEAR COMBINATIONS
Filed Dec. 12, 1963 5 Sheets-Sheet 1

Fig. 1

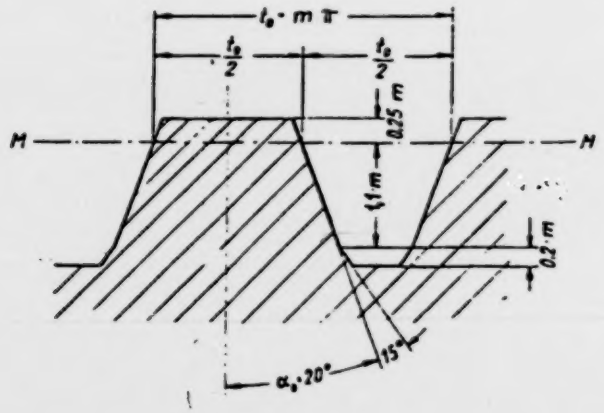
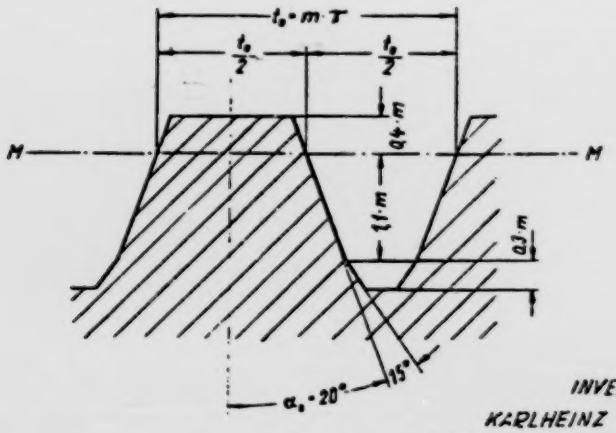


Fig. 2



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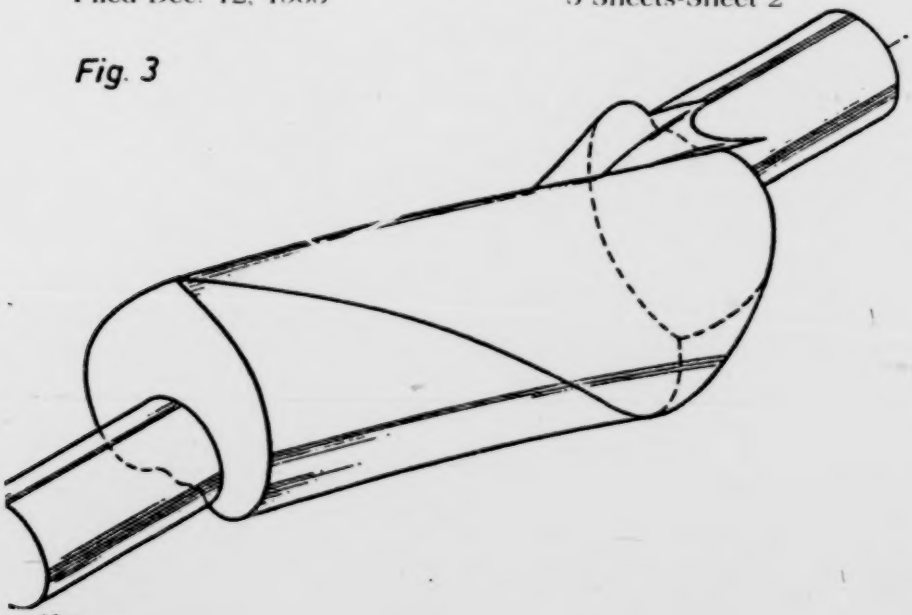
April 26, 1966 K. ROTH 3,247,736
INVOLUTE GEAR COMBINATIONS
Filed Dec. 12, 1963 5 Sheets-Sheet 2

Fig. 3



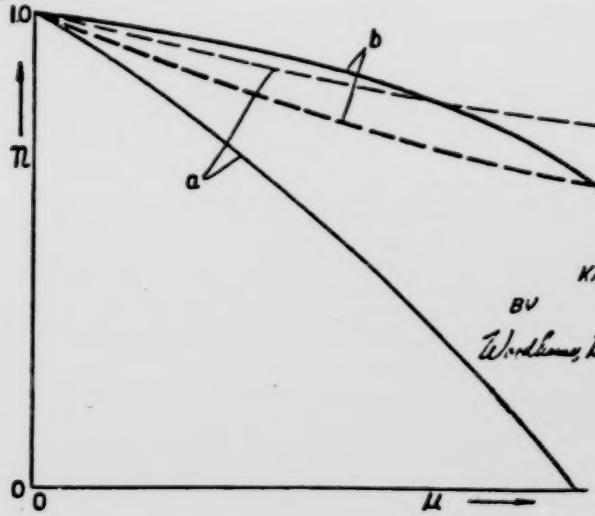


Fig. 8

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INVOLUTE GEAR COMBINATIONS
Filed Dec. 12, 1963 5 Sheets-Sheet 3

Fig. 4

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INVOLUTE GEAR COMBINATIONS
Filed Dec. 12, 1963 5 Sheets-Sheet 4

Fig. 5

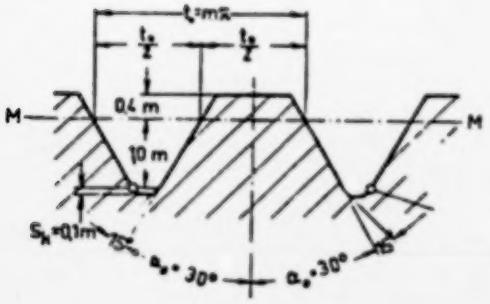
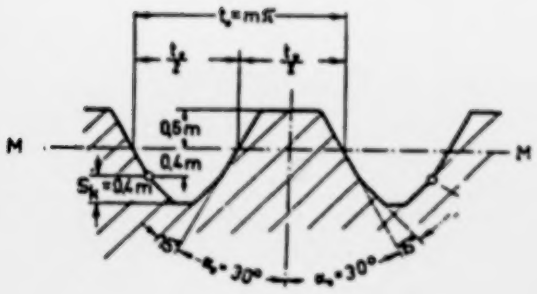


Fig. 6



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April 26, 1966 K. ROTH 3,247,736
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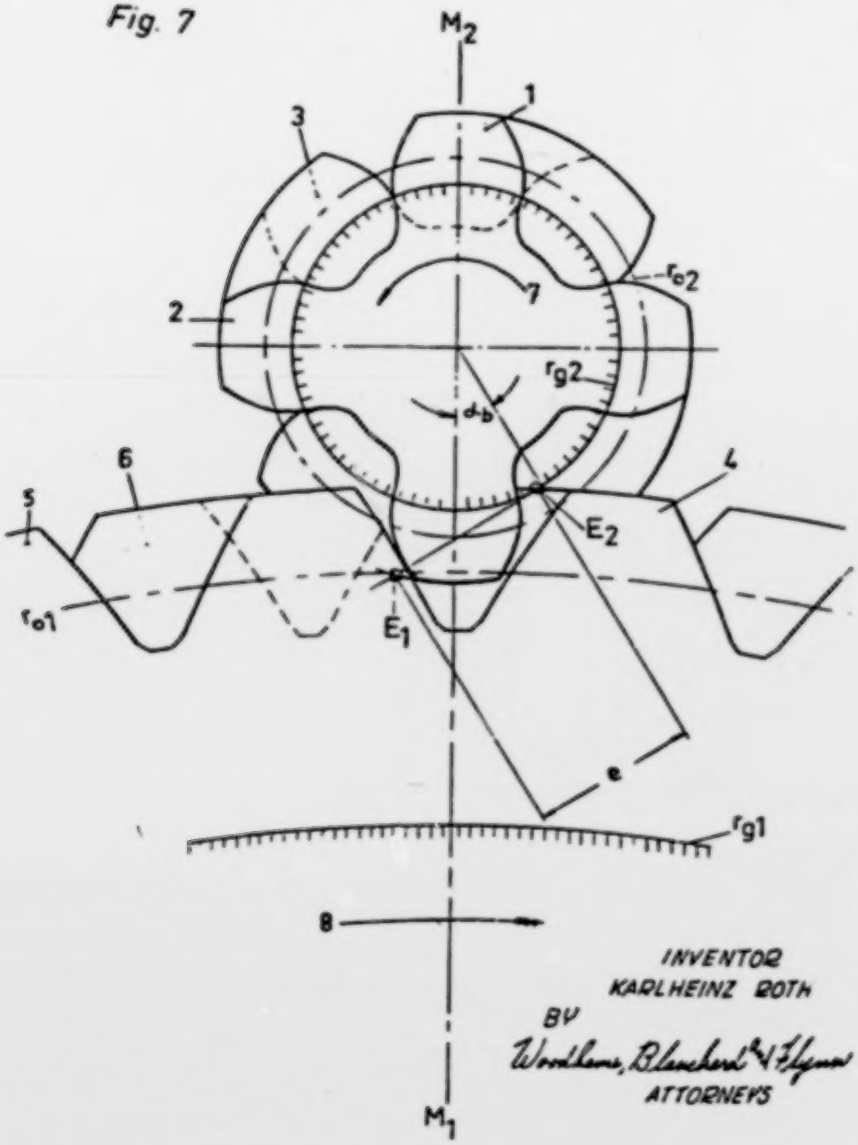
Filed Dec. 12, 1963 5 Sheets-Sheet 5

Fig. 7



INVENTOR
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United States Patent Office

3,247,736

Patented Apr. 26, 1966

1

3,247,736
INVOLUTE GEAR COMBINATIONS

Karlheinz Roth, 4 Konrad-Celtis-Strasse, Munich, Germany

Filed Dec. 12, 1963, Ser. No. 330,130
Claims priority, application Germany, Jan. 17, 1963, S 83,289
5 Claims. (Cl. 74—462)

The invention relates to a combination of involute gears wherein the pinion has a small number of teeth, as hereinafter defined.

In precision work it is particularly desirable to use gears having only a small number of teeth. The cost of making a gear of given dimensions and given quality depends on the number of teeth; the smaller the number of teeth, the lower is the cost of production. If the number of teeth on the pinion is reduced then, for a constant gear ratio, the number of teeth on the wheel is reduced to the same extent. Thus, if only two teeth are used for the pinion instead of eight, the number of teeth on the wheel is also reduced by the factor 4. The centre distance, the pitch circles and thus the addendum circles and the cross-section of the gear casing are reduced to the same extent. Further, a low number of teeth is desirable for a gear combination which is required to achieve a high step-down or step-up ratio in the fewest possible stages. Especially for this purpose a reduction in the number of teeth for the pinion leads to a further reduction in the overall size of the stage.

The invention relates to a combination of involute gears wherein the virtual number of teeth on the pinion is no more than five. The virtual number of teeth z_v, also known as the normal number of teeth, is defined by the equation

$$z_v = z/\cos^2 \beta_g \cdot \cos \beta_0$$

where z is the number of teeth in normal section, β_g is the helix angle on the base cylinder and β_0 is the helix angle on the pitch cylinder. In the gearing art, involute gears are preferred because they can be made with straight-sided cutters and any variation in the centre distance of the gears does not alter the momentary gear ratio as it would, for example, in cycloidal and spiral gear combinations. In worm gear and helical gear combinations, the axes intersect. Such gearing is therefore unsuitable where parallel axes are desired, for example in plate constructions. Even if intersecting axes bring about no structural disadvantages, the transmission efficiency of such gearing, for example worm gears, is generally lower than the efficiency of involute gears. A further disadvantage of worm gears is that step-up transmission is often impossible.

The invention aims to provide involute gearing with parallel gear axes, wherein the number of teeth on the pinion is so small that gear ratios are obtainable in one stage equal to those possible with worm gearing and without the overall size of the involute gears being much greater than that with worm gears. The gear combination according to the invention is suitable for pinion teeth in which the number z_v is equal to or less than 5 and the gear combination comprises the following features:

(a) Both gears helical, the helix angle being defined by

$$-0.35z_v + 1.4 < \frac{b}{m_n} \sin \beta_0 < 1.5$$

wherein β_0 = helix angle at pitch cylinder,

m_n = module

b = minimum width of tooth and z_v = virtual number of teeth;

(b) The pinion has a small addendum and a large effective dedendum if it is the driver or a large addendum and small useful dedendum if driven, the profiles of the gear combination being complementary in respect of the tooth sides involved in power transmission;

(c) There is a profile deviation which is positive for the pinion in a step-down combination but negative in a step-up combination and positive for the wheel in a step-up combination.

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As a result of an adequate profile engagement factor obtained by means of the invention, the required helix angle β_0 is much smaller than hitherto.

The reduction in the addendum or dedendum, as the case may be, and the profile deviation, may be adapted to one another so that the greater part of the line of action (relatively to the centre line and to the driver) lies on the side adjacent the point of final engagement. The pressure angle (half the angle of the sides of the basic rack profile) is preferably from 15° to 25° for a step-down combination and from 25° to 35° for a step-up combination. Further, in a step-down involute gear combination according to the invention it is preferred to use as the basic profile for the wheel the standard profile for precision work as laid down by German Industrial Standard No. 58400 (draft September 1963).

Three examples of the invention are illustrated in the accompanying drawing, wherein:

FIG. 1 is the basic rack profile in normal section for a pinion having a single tooth;

FIG. 2 is the basic rack profile in normal section for a pinion having two teeth;

FIG. 3 is a perspective view of a pinion having a single tooth;

FIG. 4 is a perspective view of the pinion having two teeth;

FIGS. 5 and 6 illustrate the two complementary basic profiles for a step-up combination;

FIG. 7 illustrates a step-up combination, and

FIG. 8 is a graph showing the basic course of the efficiency of the step-up gearing according to FIG. 7.

In the profile according to FIG. 1, the module m is, as usual, taken as the reference parameter. All the measurements except the angles will be in millimetres when multipled by the module. The chain dotted line M—M is located so that the tooth thickness is equal to the tooth gap and equal to half the pitch t_0. The pitch t_0 amounts to 3.14159 modules. The addendum is 0.25 module, the dedendum 1.1 module and the bottom clearance 0.2 module.

A helical gear made with these dimensions by a hob milling process is illustrated in FIG. 3. The face width amounts to about 6 modules; the helix angle at the pitch cylinder $\beta_0 = 15.2°$. By using the helical tooth form, a transverse contact ratio larger than 1 is obtainable for the gear combination as a whole even though the transverse contact ratio in any one lamina plane of the combination is less than 1. The minimum helix angle necessary for engagement is limited by the formula

$$-0.35z_v + 1.4 < \frac{b}{m_n} \sin \beta_0 < 1.5$$

In order to keep the axial forces on the gears to a minimum, a small helix angle is desirable. In addition, a small helix angle has a favourable effect on the form of the sides of the gears if they are made with a hob milling machine. The spread, i.e. the arc of the pitch circle between the two pitch cylinder lines through the points of intersection between a line passing through a side of the tooth and the two normal faces of the gearing, is chosen so that the transverse contact ratio ε of the combination is larger than 1. The mating profile can

be a standard profile, preferably according to German Industrial Standard 58400 draft September 1963 for the precision industry. For the case where the pinion has one tooth ($z=1$), the spread is half the pitch ($t_0/2$), the wheel has 30 teeth and the helix angle β_0 is 15.2°, the transverse contact ratio ϵ will be equal to or larger than 1.1.

The standard profile for the procision industry according to German Industrial Standard No. 58400 is complementary to the basic profile of FIG. 1 in respect of the sides of the teeth involved in power transmission, i.e. the pinion of FIGS. 1 or 3 has a small addendum and a large dedendum, whilst the mating wheel has a large addendum and a small dedendum as far as the portions involved in power transmission are concerned. Thus, in the case of the standard profile according to German Industrial Standard 58400 draft September 1963, a portion of the dedendum available for power transmission is not used in the gear combination in question. This unutilised portion can, however, be utilised in co-operation with another gear. Naturally, it is also possible to provide a basic profile which has a small dedendum to begin with. Such a gear has the advantage of being somewhat stronger but the disadvantage of being useful only in combination with the pinion of FIGS. 1 or 3.

The basic profile of FIG. 2 has, in relation to that of FIG. 1, a somewhat larger addendum, namely 0.4 module, and a somewhat larger bottom clearance of 0.3 module. The mating pinion (FIG. 4) likewise has a face advance of half the pitch ($t_0/2$). The minimum face width is obtained by the formula

$$-0.35z_v+1.4<\frac{b}{m_n}\sin\beta_0<1.5$$

The helix angle is 15.2°. The pinion of FIG. 4 can likewise be combined with a helical gear made according to German Industrial Standard 58400 draft September 1963. The basic profile of FIG. 2 is also complementary to the profile of German Standard 58400 draft September 1963. Further, it is possible, although less advantageous to use a mating helical gear according to German Industrial Standard No. 867.

FIG. 5 shows the basic profile for a gear in a step-up combination. m represents the module, to t_0 the pitch, $\alpha_0=30°$ half the angle of the side, $S_K=0.1\ m$ the bottom clearance. The reference line M—M represented by a chain dotted line is located so that the tooth thickness and tooth gap are equal. The point P designates the end of the useful tooth surface of the basic rack profile. FIG. 6 illustrates a basic profile complementary to the basic profile of FIG. 5 and the same references are used as in FIG. 5.

A gear combination consisting of helical gears corresponding to the basic profiles of FIGS. 5 and 6 is illustrated in FIG. 7. The pinion 1 has four teeth and the wheel 4 has forty teeth. The drawing shows the front faces 2 and the rear faces 3 for the pinion 1 and the front faces 5 and rear faces 6 for the wheel 4. The transverse contact ratio for a laminate amounts to about 0.7 and when considering the face contact ratio the total engagement factor is 1.2. The direction of rotation of the gears 1 and 4 is represented by the arrows 7 and 8, respectively; thus, the illustrated version is a step-up gearing. The line of action E1, E2 is located principally on the righthand side of the line M1, M2, i.e. on the side adjacent the point of final engagement. Upon commencement of engagement (pushing friction) the sides of the teeth slide on one another only to a small extent because the point E1 lies close to the centre line M1, M2, whereas upon termination of engagement (pulling friction) at the point E2 the more intensive sliding of the tooth sides upon one another is less detrimental. FIG. 8 is a graphical representation of the momentary efficiency relatively to the coefficient of friction μ for a conventional gear combination (a) and for a laminate

of the gear combination according to the invention (b). Although the number of teeth in the combination according to the invention is less, the efficiency is higher.

The curves for commencement of engagement are shown in full lines whilst the curves for termination of engagement are shown in broken lines. As evident from FIG. 8, the curves for the gear combination according to FIG. 7 lie closer to one another than the corresponding curves for conventional gear combinations. In particular, the drop in efficiency η upon an increase in friction μ is not so great upon commencement of engagement and always lies considerably above zero. The principal advantage of this is that jamming of the gears upon commencement of engagement is avoided with the construction according to the invention even with a coefficient of friction $\mu=1.0$. Accordingly, the gear combination according to the invention is particularly suitable for difficult operating conditions in which the coefficient of friction may become particularly high as a result of infrequent servicing and/or the presence of sand or dust in the atmosphere. The graphs for a lamina normal section also apply in this case.

The general limitations for the helix angle β_0 have been mentioned previously. In a step-down combination

$$-0.35z_v+1.4<\frac{b}{m_n}\sin\beta_0<-0.4z_v+2,$$

whereas in a step-up combination

$$0.6<\frac{b}{m_n}\sin\beta_0<1.5$$

For the profile deviation x of the pinion

$$0<x<-0.35z_v+2.5$$

in a step-down combination and

$$x<0$$

in a step-up combination and in general

$$-0.125z_v+0.2<x<-0.35z_v+2.5$$

The following are practical values for the profile deviation.

Step-down gearing:

- Five teeth on pinion, $x=+0.7$
- Four teeth on pinion, $x=+0.88$
- Three teeth on pinion, $x=+0.92$
- Two teeth on pinion, $x=+0.98$
- One tooth on pinion, $x=+1.03$

Step-up gearing:

- Three teeth on pinion, $x=-0.2$ and thus $x=+0.3$ for the wheel
- Four teeth on pinion, $x=-0.15$ and thus $x=+0.5$ for the wheel
- Five teeth on pinion, $x=-0.22$ and thus $x=+0.5$ for the wheel

The following values are examples for the addendum and dedendum in step-down gear combinations.

Addendum for the driver, useful dedendum for the driven:

One tooth on pinion	0.25
Two teeth on pinion	0.4
Three teeth on pinion	0.5
Four teeth on pinion	0.6
Five teeth on pinion	0.7

The addendum of the driven gear and the corresponding dedendum of the driving gear may be 1.1 in this case.

For step-down gear combinations the driving gear may have an addendum of 0.4 and a dedendum of 1.0, without any modifications.

The addendum as referred to above is given by

$$h_k=\tfrac{1}{2}[d_k-(z+2x)m]$$

wherein d_k=diameter of addendum circle, z=number of teeth, x=profile deviation factor and m=module.

The profile deviation xm is the displacement of the reference line of the cutting tool from the pitch circle of the wheel. The reference line of the cutting tool is located where the tooth thickness is equal to the tooth gap.

What I claim is:

1. A combination of involute gears in which the shafts of the two gears are parallel to one another and wherein the virtual number of teeth on the pinion is no more than five, comprising the following features:

(a) Both gears are helical and the helix angle is given by

$$-0.35z_v+1.4<\frac{b}{m_n}\sin\beta_0<1.5$$

wherein β_0=helix angle at the pitch cylinder, m_n=module, b=minimum tooth width and z_v=virtual number of teeth;

(b) the pinion has a small addendum and a large useful dedendum (difference between useful height of tooth and addendum) if it is the driver, or a large addendum and a small useful dedendum if it is the driven, the profiles of the combination being complementary in respect of the tooth sides involved in power transmission;

(c) there is a profile deviation which is positive for the pinion in a step-down combination but negative in a step-up combination, and positive for the wheel in a step-up combination.

2. A combination according to claim 1 wherein the reduction in the addendum or dedendum, as the case may be, and the profile deviation are adapted to one another so that the greater part of the line of action (relatively to the centre line and relatively to the driver) lies on the side adjacent final engagement.

3. A step-down gear combination according to claim 1 wherein the pressure angle (half the angle of the sides of the basic rack profile) is between 15° and 25°.

4. A step-up gear combination according to claim 1 in which the pressure angle is between 25° and 35°.

5. A step-down gear combination according to claim 1 wherein the basic profile of the wheel is the standard profile for the precision industry according to German Industrial Standard No. 58400 draft September 1963.

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DON A. WAITE, *Primary Examiner.*

TRANSLATION OF GERMAN APPLICATION
filed January 17, 1963
copied from
DX 26, Appendix Vol. IV, pp. 1703-09

Involute gear mating with a small number of teeth on the smaller gear

It is especially desirable for precision engineering to get along with small numbers of teeth on the gears. The production costs of a gear of given dimensions and given quality depend on the number of the individual teeth; the smaller the number of teeth, the lower the cost of the individual gear will be. If one decreases the number of the teeth on the smaller gear (pinion gear), then the number of teeth on the bigger gear is reduced to the same degree while maintaining the same gear ratio; thus, if instead of eight only two teeth are used for the pinion gear, then the number of teeth on the bigger gear is reduced by the factor of 4. A reduction in the number of teeth is, moreover, desirable in the case of a mating that is to provide a big step-down or step-up. Above all, for this type of application the reduction in the number of teeth on the smaller gear leads to a further reduction in the overall structural size of the stage.

United States Patent Office

3,247,376
Patented Apr. 26, 1966
1
3,247,736

INVOLUTE GEAR COMBINATIONS

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Filed Dec. 12, 1963, Ser. No. 330,130
Claims priority, application Germany, Jan. 17, 1963, S 83,289
5 Claims. (Cl. 74—462)

The invention relates to a combination of involute gears wherein the pinion has a small number of teeth, as hereinafter defined.

In precision work it is particularly desirable to use gears having only a small number of teeth. The cost of making a gear of given dimensions and given quality depends on the number of teeth; the smaller the number of teeth, the lower is the cost of production. If the number of teeth on the pinion is reduced then, for a constant gear ratio, the number of teeth on the wheel is reduced to the same extent. Thus, if only two teeth are used for the pinion instead of eight, the number of teeth on the wheel is also reduced by the factor 4. The centre distance, the pitch circles and thus the addendum circles and the cross-section of the gear casing are reduced to the same extent. Further, a low number of teeth is desirable for a gear combination which is required to achieve a high step-down or step-up ratio in the fewest possible stages. Especially for this purpose a reduction in the number of teeth for the pinion leads to a further reduction in the overall size of the stage.

Translation of
German Application

The invention concerns an involute gear mating with a pinion gear tooth number $Z1 \leq 5$. =

As is known, involute gears are preferred in transmission technology because they can be produced with straight-flanked tools and a change in the axial distance of the gears does not change the instantaneous transmission ratio, as is the case, e.g., with cycloidal and circular arc gear mating. As is known, the axes intersect in worm gear and helical gear mating. These gear systems are therefore not suitable for gearing designs with parallel axes, e.g., for those executed in Platinenbauweise. The transmission efficiency of a worm gear mating is usually worse than the efficiency of an involute gear mating; a further disadvantage of the worm gear transmission is that a step-up is not possible. It is the problem of the invention to create an involute gear mating for parallel axes of the gears in which the number of teeth on the smaller gear (pinion) is so small that with the involute gearing matings according to the invention, transmissions can be realized in one stage which equal the possible transmissions of a worm drive without the strucutral size of the involute gear mating according to the invention fundamentally exceeding that of the worm drive. The gear mating of the invention is suitable for pinion gear tooth numbers under 5 and is characterized by the following features: =

U. S. Patent

The invention relates to a combination of involute gears wherein the virtual number of teeth on the pinion is no more than five. The virtual number of teeth z_v, also known as the normal number of teeth, is defined by the equation

$$z_v = z/\cos^2 \beta_g \cdot \cos \beta_0$$

Added Disclosure

where z is the number of teeth in normal section, β_g is the helix angle on the base cylinder and β_0 is the helix angle on the pitch cylinder. In the gearing art, involute gears are preferred because they can be made with straight-sided cutters and any variation in the centre distance of the gears does not alter the momentary gear ratio as it would, for example, in cycloidal and spiral gear combinations. In worm gear and helical gear combinations, the axes intersect. Such gearing is therefore unsuitable where parallel axes are desired, for example in plate constructions. Even if intersecting axes bring about no structural disadvantages, the transmission efficiency of such gearing, for example worm gears, is generally lower than the efficiency of involute gears. A further disadvantage of worm gears is that step-up transmission is often impossible.

The invention aims to provide involute gearing with parallel gear axes, wherein the number of teeth on the pinion is so small that gear ratios are obtainable in one stage equal to those possible with worm gearing and without the overall size of the involute gears being much greater than that with worm gears. The gear combination according to the invention is suitable for pinion teeth in which the number z_v is equal to or less than 5 and the gear combination comprises the following features:

Translation of
German Application

a) Such profiles are used for the smaller gear as have for the drive gear a small crown of tooth height and a big useful root of tooth depth, for the driven gear a big crown of tooth height and small useful root of tooth depth; the profiles of the tooth mating behave complementarily to each other with reference to the flank involved in the power transmission. =

b) A profile displacement is made such that for a gear mating for stepping down a rotary motion the smaller gear (pinion) has a positive profile displacement, for a gear mating for stepping up, on the other hand, the smaller gear (pinion) has a negative profile displacement. =

c) Crown shortening or root of tooth decrease and profile displacement are so adjusted to each other that the length of contact (with reference to the center line and with reference to the driving gear) for the most part lies on the side of the contact end. =

d) The contact angle α_0 is selected for a gear mating for stepping down of 15° to 25°, while for a gear mating for stepping up a contact angle of 25° to 35° is selected. =

U. S. Patent

(a) Both gears helical, the helix angle being defined by

$$-0.35z_v + 1.4 < \frac{b}{m_n} \sin \beta_o < 1.5$$

wherein β_o = helix angle at pitch cylinder,

m_n = module

b = minimum width of tooth and z_v = virtual number of teeth;

Additional Disclosure

(b) The pinion has a small addendum and a large effective dedendum if it is the driver or a large addendum and small useful dedendum if driven, the profiles of the gear combination being complementary in respect of the tooth sides involved in power transmission;

(c) There is a profile deviation which is positive for the pinion in a step-down combination but negative in a step-up combination and positive for the wheel in a step-up combination.

As a result of an adequate profile engagement factor obtained by means of the invention, the required helix angle β_o is much smaller than hitherto.

New

The reduction in the addendum or dedendum, as the case may be, and the profile deviation, may be adapted to one another so that the greater part of the line of action (relatively to the centre line and to the driver) lies on the side adjacent the point of final engagement. The pressure angle (half the angle of the sides of the basic rack profile) is preferably from 15° to 25° for a step-down combination and from 25° to 35° for a step-up combination. Further, in a step-down involute gear combination according to the invention it is preferred to use as the basic profile for the wheel the standard profile for precision work as laid down by German Industrial Standard No. 58400 (draft September 1963).

New

Translation of
German Application

The invention is discussed with reference to the drawing in three embodiments.

Fig. 1 shows the reference profile of the normal section (rack profile) of a one-toothed pinion,

Fig. 2 shows the reference profile of the normal section (rack profile) of a two-toothed pinion,

Fig. 3 shows a one-toothed pinion in perspective,

Fig. 4 shows a two-toothed pinion in perspective,

Fig. 5 and Fig. 6 show the reference profile of two complementary profiles for stepping up,

Fig. 7 shows a gear mating for stepping up,

Fig. 8 shows an efficiency diagram of the transverse section for contact beginning and contact end.

In the profile of Fig. 1 the module m, as usual, is taken as standard; all dimensions except for the degree designations are therefore to be multiplied with the module in mm. The line MM is so located that the tooth thickness = the space width = the half pitch $t_o/2$. The pitch t_o amounts to 3.14159 module. The tooth crown is 0.25 module high, the root is 1.1 module deep; the tip clearance is 0.2 module.

A gear helical-toothed according to these measurements in the hob method is shown in Fig. 3. The tooth width is about 6 module, the angle of skew on the pitch cylinder $\beta_0 = 15.2°$.

The smallest angle of skew necessary for the *Ueberwachung* (monitoring, control, inspection??) is calculated according to the formula $\sin \beta = \pi m/2b$ (m = module, b = tooth width);

U. S. Patent

Three examples of the invention are illustrated in the accompanying drawing, wherein:

FIG. 1 is the basic rack profile in normal section for a pinion having a single tooth;

FIG. 2 is the basic rack profile in normal section for a pinion having two teeth;

FIG. 3 is a perspective view of a pinion having a single tooth;

FIG. 4 is a perspective view of the pinion having two teeth;

FIGS. 5 and 6 illustrate the two complementary basic profiles for a step-up combination;

FIG. 7 illustrates a step-up combination, and

FIG. 8 is a graph showing the basic course of the efficiency of the step-up gearing according to FIG. 7.

In the profile according to FIG. 1, the module m is, as usual, taken as the reference parameter. All the measurements except the angles will be in millimetres when multipled by the module. The chain dotted line M—M is located so that the tooth thickness is equal to the tooth gap and equal to half the pitch t_o. The pitch t_o amounts to 3.14159 modules. The addendum is 0.25 module, the dedendum 1.1 module and the bottom clearance 0.2 module.

A helical gear made with these dimensions by a hob milling process is illustrated in FIG. 3. The face width amounts to about 6 modules; the helix angle at the pitch cylinder $\beta_o = 15.2°$. By using the helical tooth form, a transverse contact ratio larger than 1 is obtainable for the gear combination as a whole even though the transverse contact ratio in any one lamina plane of the combination is less than 1. The minimum helix angle necessary for engagement is limited by the formula

$$-0.35z_v + 1.4 < \frac{b}{m_n} \sin \beta_o < 1.5$$

Substantially different, and more definite.

in order to keep the axial force of the gears small, a small angle of skew is desirble; such a one also works out favorably on the flank shape of the gears in the production with gear hobbers.The lead, i.e., the pitch circle arc between the two pitch cylinder generatrices through the points of intersection of a flank line with the two faces of the toothing, is so chosen with a ratio of lead of the mating of $\epsilon > 1$ results. As mating profile a standard profile can be used and with particularly good results the 58 400 for precision engineering; here, with a lead of a half pitch $t_o/2$ on a gear with 30 teeth and a spiral angle of $\beta_0 = 15.2°$ a lead ratio of $\epsilon = 1.1$ results.

Precision engineering standard profile 58400 behaves, with reference to the flanks involved in the power transmission, compelentarily to the reference profile of Fig. 1, i.e., the pinion gear of Fig. 1 or 3 has a small crown and a large root, while the gear meshing with this pinion relative to parts involved in the power transmission has a large crown and a small root. Thus in standard profile 58400 a part of the flank of the root available for the power transmisssion in the gear mating in question is not utilized; this unused part can, however, be made useful if used with an additional gear. Of course, it is also possible to provide a reference profile that has a small root to begin with. Such a gear would have the advantage of the somewhat greater solidity and the disadvantage that it can be mated only with the pinion gear as per Fig. 1 or 3.

In order to keep the axial forces on the gears to a minimum, a small helix angle is desirable. In addition, a small helix angle has a favourable effect on the form of the sides of the gears if they are made with a hob milling machine. The spread, i.e. the arc of the pitch circle between the two pitch cylinder lines through the points of intersection between a line passing through a side of the tooth and the two normal faces of the gearing, is chosen so that the transverse contact ratio ϵ of the combination is larger than 1. The mating profile can be a standard profile, preferably according to German Industrial Standard 58400 draft September 1963 for the precision industry. For the case where the pinion has one tooth ($z = 1$), the spread is half the pitch ($t_o/2$), the wheel has 30 teeth and the helix angle β_o is 15.2°, the transverse contact ratio ϵ will be equal to or larger than 1.1.

The standard profile for the procision industry according to German Industrial Standard No. 58400 is complementary to the basic profile of FIG. 1 in respect of the sides of the teeth involved in power transmission, i.e. the pinion of FIGS. 1 or 3 has a small addendum and a large dedendum, whilst the mating wheel has a large addendum and a small dedendum as far as the portions involved in power transmission are concerned. Thus, in the case of the standard profile according to German Industrial Standard 58400 draft September 1963, a portion of the dedendum available for power transmission is not used in the gear combination in question. This unutilised portion can, however, be utilised in co-operation with another gear. Naturally, it is also possible to provide a basic profile which has a small dedendum to begin with. Such a gear has the advantage of being somewhat stronger but the disadvantage of being useful only in combination with the pinion of FIGS. 1 or 3.

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The reference profile of Fig. 2 has a somewhat larger crown than that of Fig. 1, namely, of 0.4 module and a somewhat greater tip clearance (root radius tooth fillet) of 0.3 module. The therewith associated pinion gear (Fig. 4) likewise has a lead of a half pitch $t_o/2$, the minimum tooth width being about 6 module;

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the spiral angle is 15.2°. The pinion of Fig. 4 can likewise be mated with a helical gear that is produced according to precision engineering standard profile 58400. The reference profile of Fig. 2 is likewise a complementary profile to precision engineering standard profile 58400. It is possible, though a little less advantageous, to use a helical gear according to DIN 867 as mating gear.

Fig. 5 shows a reference profile for gearing for stepping up a rotary motion. In it, m designates the module, to the pitch $\alpha_0 = 30$, the pressure angle $S_K = 0.1$ m the tip clearance. The profile reference line MM, indicated by dash-dot line, is so located that tooth thickness and tooth space are equal. The point P marks the end of the driving flank of the rack profile. Fig. 6 shows a reference profile complementary to the reference profile of Fig. 5. The designations are the same as in Fig. 5.

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U. S. Patent

The basic profile of FIG. 2 has, in relation to that of FIG. 1, a somewhat larger addendum, namely 0.4 module, and a somewhat larger bottom clearance of 0.3 module. The mating pinion (FIG. 4) likewise has a face advance of half the pitch ($t_O/2$). The minimum face width is obtained by the formula

$$-0.35z_v + 1.4 < \frac{b}{m_n} \sin \beta_o < 1.5$$

U.S. case substantially augmented to provide specific data.

The helix angle is 15.2°. The pinion of FIG. 4 can likewise be combined with a helical gear made according to German Industrial Standard 58400 draft September 1963. The basic profile of FIG. 2 is also complementary to the profile of German Standard 58400 draft September 1963. Further, it is possible, although less advantageous to use a mating helical gear according to German Industrial Standard No. 867.

FIG. 5 shows the basic profile for a gear in a step-up combination. m represents the module, to t_o the pitch, $\alpha_o = 30°$ half the angle of the side, $S_K = 0.1\ m$ the bottom clearance. The reference line M—M represented by a chain dotted line is located so that the tooth thickness and tooth gap are equal. The point P designates the end of the useful tooth surface of the basic rack profile. FIG. 6 illustrates a basic profile complementary to the basic profile of FIG. 5 and the same references are used as in FIG. 5.

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A gear mating composed of helical gears corresponding to the reference profile of Figs. 5 and 6 is shown in Fig. 7. The smaller gear (pinion gear 1) has six teeth, while the gear 4 has 60 teach. Of individual teeth of pinion gear 1 front surfaces 2 and back surfaces 3 are indicated, of individual teeth of the gear 4 front surfaces and back surfaces 6 are indicated. The ratio of lead of a *Schichtelement* (layer/coating element?) is 0.7, taking into consideration the ratio of lead, the entire overlap amounts to 1.2. The direction of rotation of the gears 1 and 4 is indicated by the arrows 7 and 8; this results in a stepping-up of a rotary motion.

The length of contact E1, E2, E3 lies, with reference to the line M1, M2 for the greater part on the right side, i.e., on the side of the contact end. At the beginning of the tooth contact (shock friction) the teeth surfaces still rub only a little on each other, since the point E1 lies close to the center line M1, M2, while at the contact end (pulling friction), at point E3 the stronger sliding of the tooth surfaces on each other does less damage. In Fig. 8, the instantaneous degree of efficiency η over the friction value μ for a normal straight gear mating and for a Schichtelement of the gear mating according to the invention is shown. Although the number of teeth of the gear mating produced according to the invention is smaller, the efficiency for the transmission is better. The curve paths z1/z2 – 60/6 are valid for a *Schichtelement* of the gear mating of the invention according to Fig. 7, the curve parts z1/z2 = 80/8 are valid for a normal straight gear mating with 80 and 8 teeth respectively.

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A gear combination consisting of helical gears corresponding to the basic profiles of FIGS. 5 and 6 illustrated in FIG. 7. The pinion 1 has four teeth and the wheel 4 has forty teeth. The drawing shows the front faces 2 and the rear faces 3 for the pinion 1 and the front faces 5 and rear faces 6 for the wheel 4. The transverse contact ratio for a laminate amounts to about 0.7 and when considering the face contact ratio the total engagement factor is 1.2. The direction of rotation of the gears 1 and 4 is represented by the arrows 7 and 8, respectively; thus, the illustrated version is a step-up gearing. The line of action E1, E2 is located principally on the righthand side of the line M1, M2, i.e. on the side adjacent the point of final engagement. Upon commencement of engagement (pushing friction) the sides of the teeth slide on one another only to a small extent because the point E1 lies close to the centre line M1, M2, whereas upon termination of engagement (pulling friction) at the point E2 the more intensive sliding of the tooth sides upon one another is less detrimental. FIG. 8 is a graphical representation of the momentary efficiency relatively to the coefficient of friction μ for a conventional gear combination (a) and for a laminate of the gear combination according to the invention (b). Although the number of teeth in the combination according to the invention is less, the efficiency is higher.

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The curve paths for the contact beginning are solid, while the curves for the contact endings are dashed. As can be seen from the representation, the curve paths for the gear mating of the invention according to Fig. 7 (are) closer together than in any previous normal gear mating; above all, the decrease in the degree of efficiency under increasing friction value is not so great with reference to the contact beginning and is always far above zero. This represents a great advantage because a jamming of the gears at the beginning of contact in the embodiment of the invention does not occur even with friction value $\mu = 1.0$. It is shown, therefore, that the gearing of the invention is especially suitable for overcoming difficult operating conditions under which the friction value is especially high as a result of infrequent maintenance of the gearing or because of the presence of sand or dust in the atmosphere. Valid here also are the diagrams for a *Schichtelement* of the frontal section (*Stirhschnitt*).

The curves for commencement of engagement are shown in full lines whilst the curves for termination of engagement are shown in broken lines. As evident from FIG. 8, the curves for the gear combination according to FIG. 7 lie closer to one another than the corresponding curves for conventional gear combinations. In particular, the drop in efficiency η upon an increase in friction μ is not so great upon commencement of engagement and always lies considerably above zero. The principal advantage of this is that jamming of the gears upon commencement of engagement is avoided with the construction according to the invention even with a coefficient of friction $\mu = 1.0$. Accordingly, the gear combination according to the invention is particularly suitable for difficult operating conditions in which the coefficient of friction may become particularly high as a result of infrequent servicing and/or the presence of sand or dust in the atmosphere. The graphs for a lamina normal section also apply in this case.

The general limitations for the helix angle β_o have been mentioned previously. In a step-down combination

$$-0.35z_v + 1.4 < \frac{b}{m_n} \sin \beta_o < 0.4z_v + 2,$$

whereas in step-up combination

$$0.6 < \frac{b}{m_n} \sin \beta_o < 1.5$$

For the profile deviation x of the pinion

$$0 < x < -0.35z_v + 2.5$$

in a step-down combination and

$$x < 0$$

in a step-up combination and in general

$$-0.125z_v + 0.2 < x < -0.35z_v + 2.5$$

All this (plus lines 1-4 in Col. 5) is new. This has no counterpart in Roth's German Patent application.

No counterpart
in German Application

The following are practical values for the profile deviation.

Step-down gearing:

Five teeth on pinion, $x = +0.7$
Four teeth on pinion, $x = +0.88$
Three teeth on pinion, $x = +0.92$
Two teeth on pinion, $x = +0.98$
One tooth on pinion, $x = +1.03$

Step-up gearing:

Three teeth on pinion, $x = -0.2$ and thus $x = +0.3$ for the wheel
Four teeth on pinion, $x = -0.15$ and thus $x = +0.5$ for the wheel
Five teeth on pinion, $x = -0.22$ and thus $x = +0.5$ for the wheel

The following values are examples for the addendum and dedendum in step-down gear combinations.

Addendum for the driver, useful dedendum for the driven:

One tooth on pinion	0.25
Two teeth on pinion	0.4
Three teeth on pinion	0.5
Four teeth on pinion	0.6
Five teeth on pinion	0.7

The addendum of the driven gear and the corresponding dedendum of the driving gear may be 1.1 in this case.

For step-down gear combinations the driving gear may have an addendum of 0.4 and a dedendum of 1.0, without any modifications.

The addendum as referred to above is given by

$$h_k = \tfrac{1}{2}[d_k - (z + 2x)m]$$

wherein d_k = diameter of addendum circle, z = number of teeth, x = profile deviation factor and m = module.

All this (plus lines 1-4 in Col. 5) is new. This has no counterpart in Roth's German Patent application.

B-18

No counterpart
in German Application

The profile deviation xm is the displacement of the reference line of the cutting tool from the pitch circle of the wheel. The reference line of the cutting tool is located where the tooth thickness is equal to the tooth gap.

Continuation of extensive new addition.

What I claim is:

1. A combination of involute gears in which the shafts of the two gears are parallel to one another and wherein the virtual number of teeth on the pinion is no more than five, comprising the following features:

(a) Both gears are helical and the helix angle is given by

$$-0.35z_v + 1.4 < \frac{b}{m_n} \sin \beta_0 < 1.5$$

wherein β_0 = helix angle at the pitch cylinder, m_n = module, b = minimum tooth width and z_v = virtual number of teeth;

(b) the pinion has a small addendum and a large useful dedendum (difference between useful height of tooth and addendum) if it is the driver, or a large addendum and a small useful dedendum if it is the driven, the profiles of the combination being complementary in respect of the tooth sides involved in power transmission;

(c) there is a profile deviation which is positive for the pinion in a step-down combination but negative in a step-up combination, and positive for the wheel in a step-up combination.